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GB 2169960 A GB 0878278 A GB 0729383 A GB 0714672 A

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INT CL^5 **F02B 7/00 7/02 7/04 7/06 7/08 75/12**

(54) Compression ignition engine fuel supply control

(57) Over a lower part of the power range from tick over up to a transition point, engine power is varied by varying the quantity of the main injection and there after the main injection of fuel is supplemented by a secondary introduction of the same or a different fuel. With secondary fuel introduction the main injection timing is progressively retarded to compensate for reduced ignition delay. The secondary fuel introduction may be in the form of vapourised fuel injected onto an exhaust gas recirculation valve (Fig. 8) or a glow plug in the intake manifold (Fig. 9).

At least one drawing originally filed was informal and the print reproduced here is taken from a later filed formal copy.

The claims were filed later than the filing date within the period prescribed by Rule 25(1) of the Patents Rules 1990.

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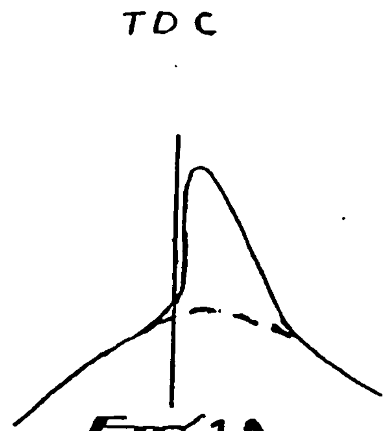


FIG. 1A

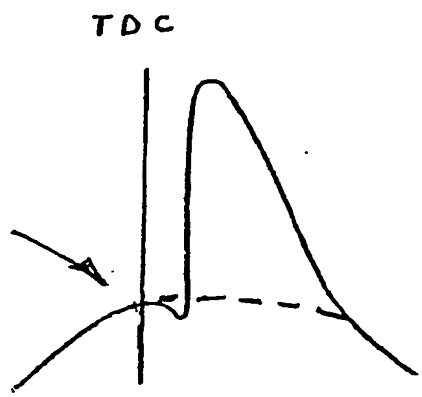


FIG. 1B

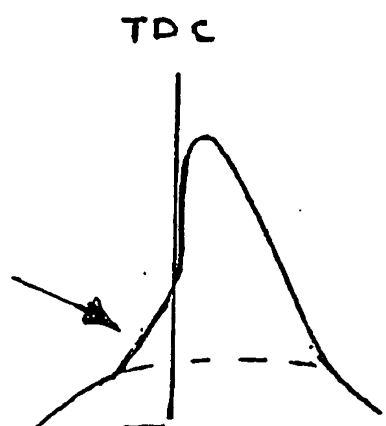


FIG. 1C

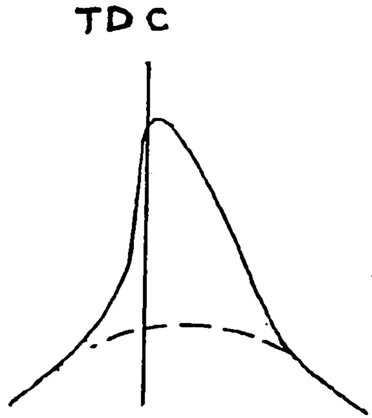


FIG. 1D

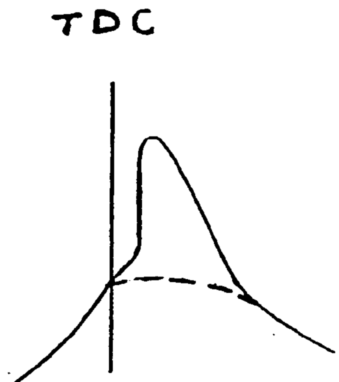


FIG. 1E

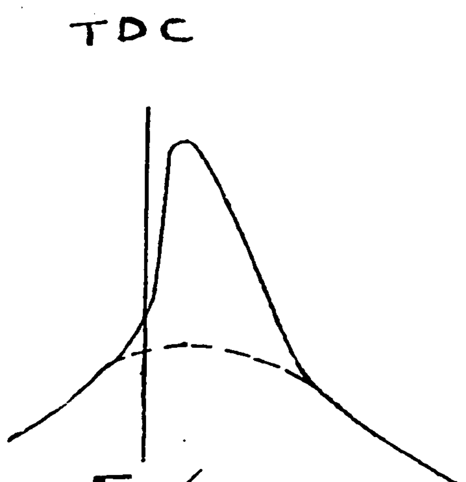


FIG. 1F

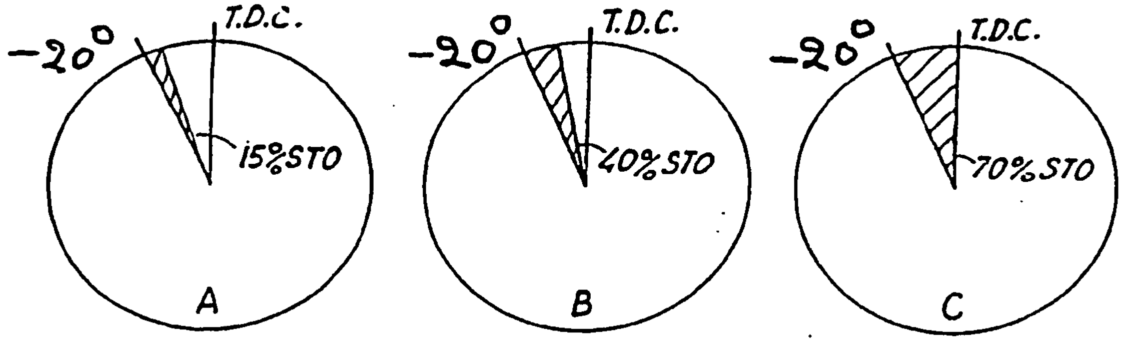


FIG. 2 (PRIOR ART)

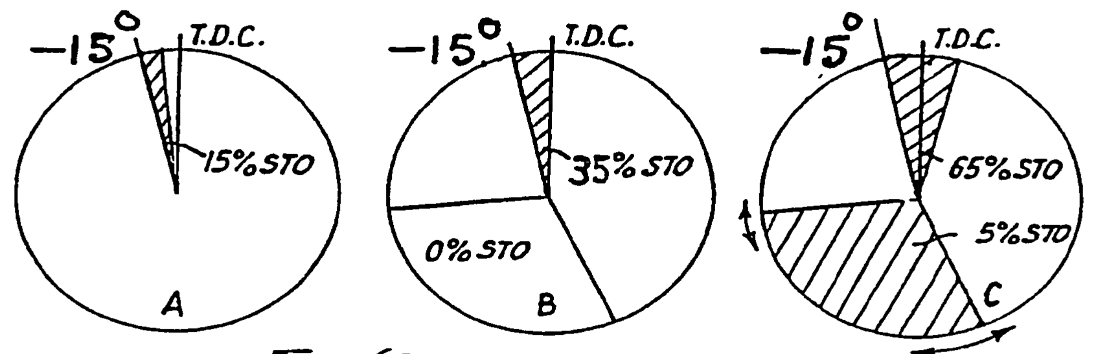
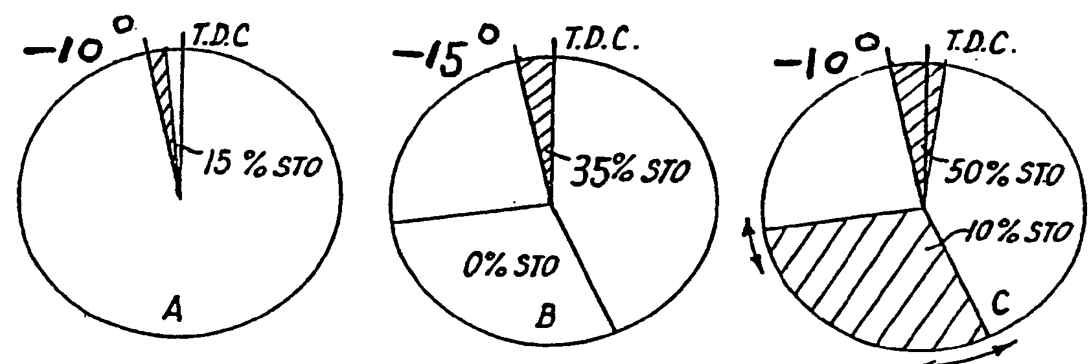


FIG. 3



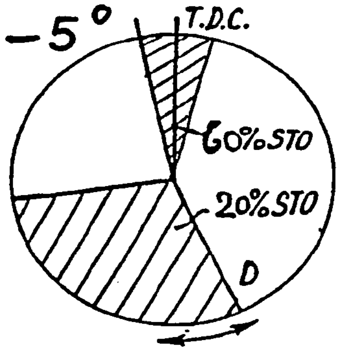


FIG. 4

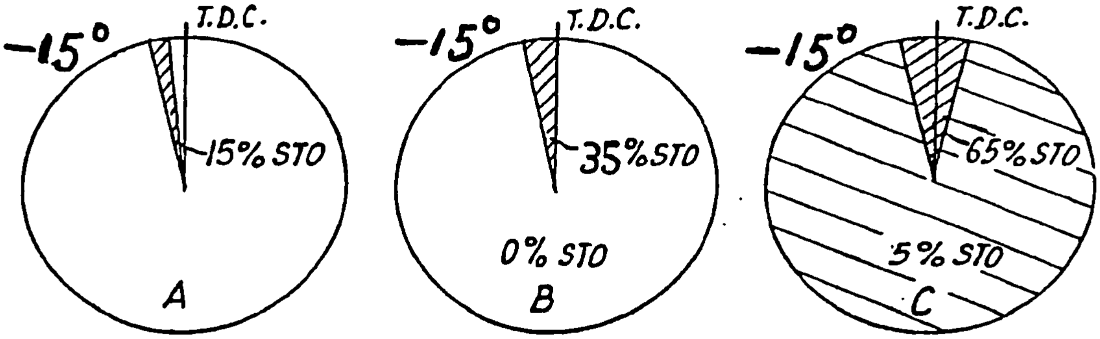


FIG. 5

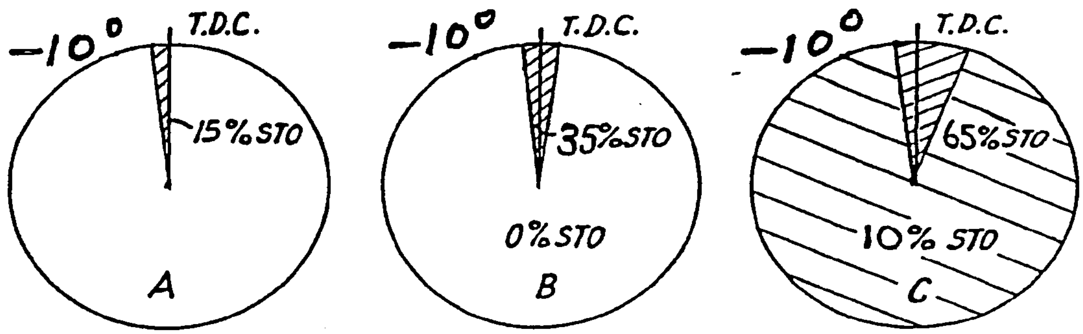


FIG. 6

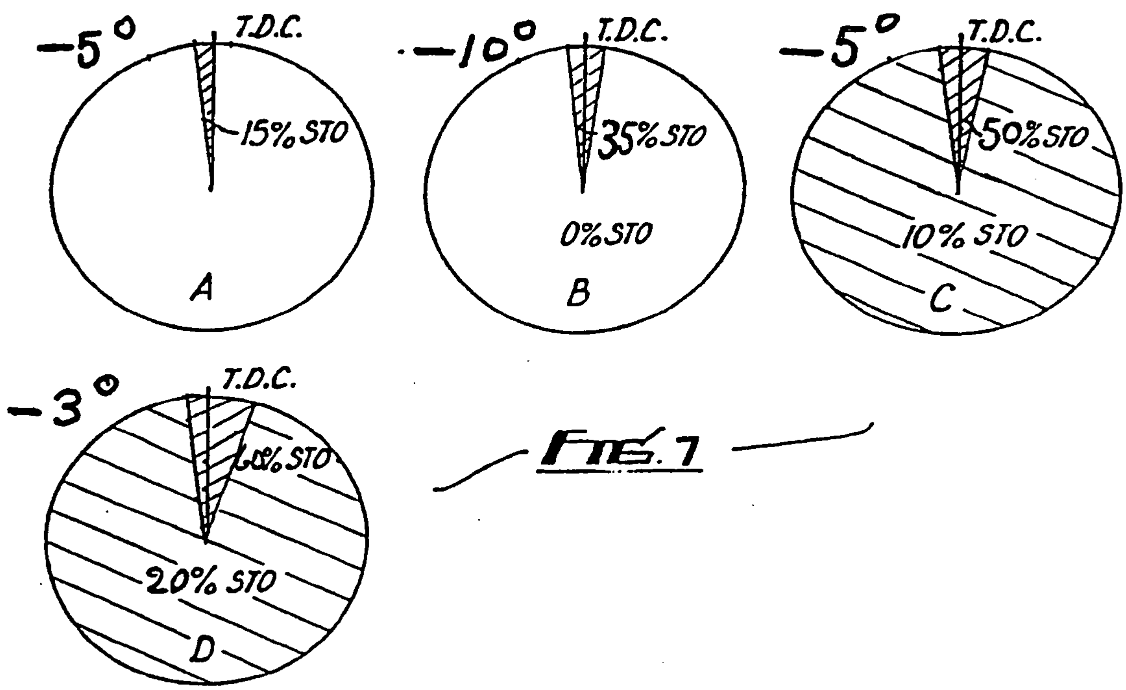


FIG. 7

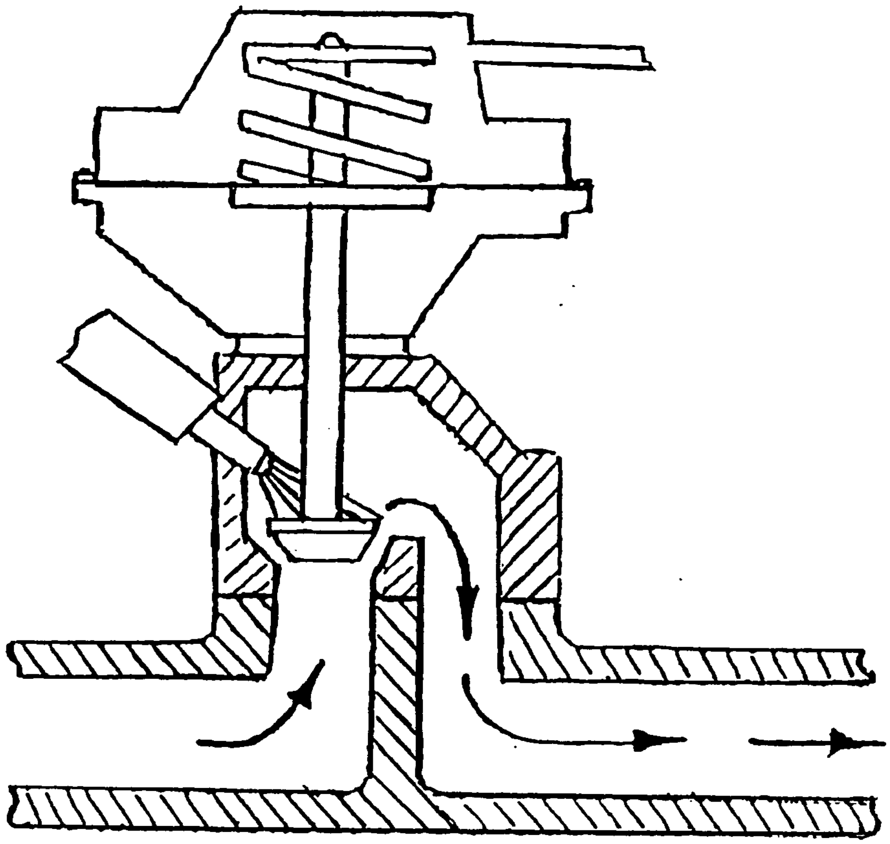


FIG. 8

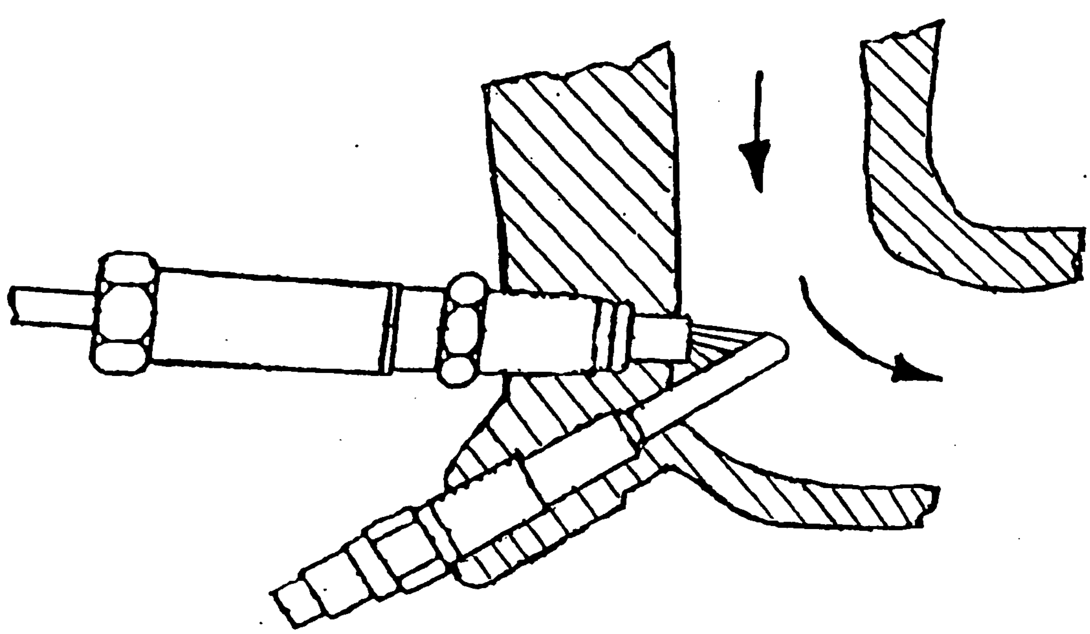


FIG. 9

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IAP17 Rec'd PCT/PTO 28 APR 2006

Compression Ignition Engine

This invention relates to the delivery of fuel to the compression ignition engine.

The excessive noise in operation and the emission of smoke from the exhaust of the diesel engine has led to the drafting of stringent legislation.

The main problem with the compression ignition engine is the noise generated at idle and acceleration and the suspension in the exhaust gases. The particles are in liquid or solid form. The liquid particles consist mainly of partially oxidised hydrocarbons and the solid particles consist of carbon from the remains of the combustion process. Although the compression ignition engine produces less nitrous oxide gas than the premixed spark ignited engine, it is desirable that this gas is kept to a minimum.

It is known that noise can be reduced by injecting less fuel into the engine before T.D.C.

Unburnt hydro-carbons come from several sources. Fuel deposited on cool cylinder walls, fuel injected at a high rate so that it defuses into a mixture which is too lean to burn and fuel that is incompletely mixed with air at the end of combustion. nitrogen oxides have been shown to be associated with the peak cylinder pressure. Peak cylinder pressure can be reduced by reducing the fuel injection before T.D.C. Particulate matter is influenced by the completeness of fuel burned. Complete combustion requires the rapid mixing of the fuel just before flammable combustion is achieved. Some of the conflicting problems are apparent such as retarding the injection timing before T.D.C will reduce noise and nitrous-oxides and advancing injection timing before T.D.C will minimise smoke and particulate. So far all diesel engine injection timings are a compromise between noise and fuel consumption and noise and smoke.

All hydro-carbon fuels when mixed with air thermo-chemically react when subjected to heat and pressure. This reaction may be small or high depending on the amount and pressure applied and the fuel's resistance to oxidation. Because chemical energy is released at different rates from different hydro-carbons the speed of ignition is influenced by the readily ignitable components of the fuel.

When diesel fuel is injected into the hot air at the end of the

compression stroke there is a high speed thermo-chemical reaction which causes the generation of heat. When this thermo-chemical reaction reaches a critical temperature flammable combustion is established. To establish flame in the diesel engine a temperature of over 1000 degrees centigrade must be reached. When flam is established self propagation of the flam is only possible when the speed of thermo-chemical heat release is sufficient to overcome any loss by radiation, convection and conduction. The speed of reaction is dependent on the temperature that the burning and burnt gases can transmit to the fuel air mixture that is about to be burned.

Although flame is established at 1000 degrees centigrade a minimum temperature of 1150 degrees centigrade must be reached to achieve a minimum speed for the self propagation of flame.

When fuel is ignited by spark ignition in the pre-mixed petrol engine cycle the minimum quantity of fuel required for self propagation of flame is governed by the overall cylinder temperature. For most pre-mixed fuels the minimum quantity required for this temperature to be achieved is over 50% chemically correct, but this amount can be reduced by higher compression temperature.

Flame will only propagate in the pre-mixed engine when the temperature of the mixture exceeds a threshold such as 1150

degrees centigrade.

This temperature threshold that can be achieved by a high temperature spark or thermo-chemical reaction will transfer thermo-chemical heat release to self-supporting flame propagation.

This temperature 1150 degrees centigrade will be referred to hereafter as the transition point. I have discovered that the transition point can be varied depending on the compression ration, ambient air temperature, and rate and timing of injection.

Once these factors determined by individual engine design are known it is relatively easy for the transition point to be established for each type of engine. As stated all hydro-carbon fuels when mixed with air thermo-chemically react when subjected to heat and pressure. There is a time lag (delay period) until the fuel has absorbed enough heat to start the chemical reaction that will commence the self generation of heat (ignition).

The speed of ignition is influenced by the readily ignitable components of the fuel and the temperature generated by compression. Diesel fuel has an ignition time of one second at 450 degrees centigrade. This rapidly reduces as the temperature rises and at 550 degrees centigrade the time for ignition is

reduced to less than 1/5th of a second.

In the high speed compression ignition engine cycle, because the time is limited, there is a minimum in cylinder temperature of about 600 degrees centigrade that must be maintained under all working conditions. When fuel is injected at tick over the quantity is so small it has no effect on the cylinder temperature. When fuel is injected at high load the quantity injected can reduce the cylinder temperature by 100 degrees centigrade and will have a considerable effect on the delay period.

Most of the heat generated by compression is generated in the last 15% of the stroke and about 9% of the total rise takes place in the final 1½% of piston movement. Because the maximum temperature is generated at, or near, T.D.C and the speed of chemical delay is in proportion to the temperature generated it is vital that injection is delayed until the last possible moment.

At 10 degrees B.T.D.C the piston is still moving upwards so that any early generation of pressure at or before this point will create an adverse resistance and will lead to a loss of power. By about 5 degrees B.T.D.C the piston has virtually stopped so that any pressure applied beyond this point will have a negligible effect on engine performance.

Although adverse pressure B.T.D.C may be small per stroke when multiplied by the numbers of cylinders and revolutions per minute it can have a considerable effect on powers and fuel consumption.

In my previous patent GB 2 169 960 I have disclosed a transition point for the introduction of secondary fuel to the compression ignition engine. I have discovered that the transition point can be varied depending on the compression ration, ambient air temperature and the quantity of secondary fuel injected.

I have further discovered that the amount of secondary fuel added will have a big influence on the main injection timing and when taken into consideration is a means of reducing noise and exhaust pollution.

Unless the introduction of secondary fuel into or with the inlet air of the engine is controlled the high degree of thermo-chemical heat (pre flam reaction) generated by compression will cause the main fuel charge to be ignited before the piston reaches T.D.C. This will cause a big drop in power and increase fuel consumption and over a given period of operation may cause considerable damage to the engine.

It is the object of the present invention to improve on my previous invention which was for the improvement of power in the compression ignition engine by the introduction of secondary fuel

to the air intake of the engine.

The invention provides a method of operating a compression ignition internal combustion engine wherein over a lower part of the power range thereof from tick over up to a transition point, engine power is varied by varying the quantity of the main injection late in the compression stroke from a minimum at tick over up to a higher value at the transition point and thereafter, over a higher part of the power ranges from the transition point up to full power, the main injection of fuel is supplemented by a secondary introduction of the same fuel, wherein the main injection is delayed when secondary introduction takes place.

The delay can be fixed over the higher power range.

The invention also provides a method of operating a compression ignition internal combustion engine wherein a secondary introduction of fuel prior to main injection is accompanied by a retarding of the main injection.

The invention further provides a compression ignition engine having means for introducing a secondary charge of fuel prior to said main injection over a higher power range of the engine and means for retarding said main injection during said higher power range.

The introduction can be by injection, by carburetter, by inclusion into the induced air, by introduction into re-cycled exhaust gas or otherwise.

Desirably the amount of retardation is varied dependent upon the amount of the secondary introduction.

The retardation can start from a large value at tick over, go to a lower value at transition, and go to a large value at full power.

I have discovered that the introduction of secondary fuel will advance ignition in proportion to the quantity of fuel added as well as reducing the transition point by the same proportion. When 5% stoichiometric secondary fuel is added to the engine the onset of ignition will be advanced by a minimum of 5 degrees. It is therefore necessary to retard the main fuel injection timing by 5 degrees. Also because 5% stoichiometric secondary fuel is added to the engine the transition point will be reduced by 5%. All the figures herein will, of course, vary slightly from engine to engine and from fuel to fuel. They are, however, broadly appropriate for most engines and fuels.

With a transition point of an engine occurring at 35% stoichiometric, secondary injection can commence when 30% stoichiometric fuel has been added by the main injector. By

delaying injection the temperature in the cylinder will be higher so that less fuel is required to maintain tick over, also the speed of ignition is accelerated and combustion becomes smoother and more complete. This will greatly reduce the knock that is associated with compression ignition engine tick over and the amount of hydro-carbon carry over to the exhaust is reduced.

With the advancement of future technology electronically controlled injection will become standard practice.

With electronic control it would be desirable to retard injection at tick over and then advance injection in unison with the quantity of main fuel injected until the transition point is reached. Beyond this point the main fuel injection would be retarded in proportion to the secondary fuel added, so that the onset of pressure rise in the cylinder would always coincide with T.D.C.

The invention will be described further, by way of example, with reference to the accompanying drawings wherein:-

Figures 1a to 1f are a series of cylinder pressure diagrams;

Figure 2 is a set of diagrams illustrating the operation of a conventional compression ignition engine;

Figure 3 is a similar set for a first preferred engine of the invention;

Figure 4 is a similar set for a second preferred engine of the invention;

Figure 5 is a similar set for a third preferred engine of the invention;

Figure 6 is a similar set for a fourth preferred engine of the invention;

Figure 7 is a similar set for a fifth preferred engine of the invention;

Figure 8 is a method of adding a secondary supply of fuel to the engine by impinging fuel on to a hot exhaust gas recirculating valve;

Figure 9 is a method of adding a charge of fuel to the engine by injecting fuel on to a flow plug in the inlet manifold.

Figure 1 illustrates a series of pressure diagrams. In the conventional compression ignition engine figures 1a to 1c there is a single main injection of fuel which commences about 20 degrees before T.D.C, at the end of the compression stroke.

After a delay of about 10 degrees the fuel gains enough heat for chemical heat release to start (ignition). Immediately after ignition commences combustion will proceed until all the fuel injected burns. At tick over as illustrated in figure 1a the quantity of fuel injected is so small it has no effect on the cylinder temperature and the pressure rise will be at or near T.D.C.

When the engine is accelerated from tick over to maximum load as illustrated in figure 1b, the quantity of fuel injected will reduce the cylinder temperature by about 100 degrees and will have a considerable effect on the delay period. This is illustrated by the dip in pressure and the delay in pressure rise in figure 1b.

Because the smoke limited power of the conventional engine is set at high temperature as soon as the temperature is reduced by the amount of fuel injected and low load operation, as applies at tick over, the engine is unable to burn all the fuel injected under high acceleration. This is because more heat is absorbed to achieve flammable combustion. The lowering of speed of combustion causes the last part of the fuel injected to burn too late in the engine cycle for combustion to be completed. This causes the smoke problem that is associates with the diesel engine when the accelerator is fully opened.

In my previous patent I have disclosed a transition point for the introduction of a secondary charge of fuel into the cylinder of the engine during the induction stroke. Although the fuel is too lean to cause premature ignition during the compression stroke, there is a high rate of thermo-chemical heating (pre-flame reaction). This release of extra heat during compression speeds up spontaneous ignition of the main charge, thereby compensating for the increased delay caused by the reduced temperature associated with acceleration from low load.

Although the introduction of secondary fuel advances ignition under initial acceleration, it can cause the main charge to ignite before the piston reaches T.D.C, and will have an adverse effect on engine efficiency. This is illustrated in figure 1c.

As acceleration continues the engine reaches its operating temperature and the timing of ignition will advance even further, as denoted in figure 1d. I have found it necessary to adjust the main injection timing in proportion to the amount of secondary fuel added. Figure 1e is a pressure diagram for an engine in accordance to a method of the invention in which the main injection of fuel is retarded by 5 degrees of tick over.

Figure 1f is a pressure diagram in accordance to a method of the invention showing a main injection of the engine retarded by 5 degrees plus the addition of 5% stoichiometric fuel added to the

engine and denotes that the maximum pressure rise takes place just after T.D.C. This eliminates the undesirable resistance to compression that is caused by premature ignition.

The present invention improves on my previous invention which was for the improvement of power in the compression ignition engine by the addition of secondary fuel to the air intake of the diesel engine. Although this method was successful for the improvement of power it was unable to achieve the stringent requirements of future exhaust pollution legislation. The present invention controls the method and means of fuel delivery to the engine so that pollution is reduced.

In the conventional compression ignition engine using diesel fuel as illustrated at figure 2 there is a single main injection of fuel which usually commences at about 20 degrees before T.D.C at the end of the compression stroke. When the air in the cylinder is hot enough to initiate spontaneous ignition. Combustion starts shortly after the commencement of injection and finishes shortly after the commencement of injection and finishes shortly after injection ceases. At full load injection continues until the smoke limiting power of the engine is reached. This point is reached by most conventional engines when the fuel/air ratio reaches 70 - 75% stoichiometric. In the conventional engine the speed of burn increases with the rising cylinder temperature.

This allows the engine to burn the fuel more efficiently when the engine reaches its maximum operating temperature.

Although the speed or burn increases with engine load the speed of reaction is controlled by the fuel's ability to penetrate the combustion gas residue left behind by the earlier combustion. This becomes more difficult as the rate of injection increases.

When exhaust emission is measured under normal injection conditions the particulate and hydro-carbon content of the exhaust gas is at a minimum value until the rate of injection reaches 65% stoichiometric. Beyond this point the particulate and hydro-carbon concentration increases in proportion to the amount of fuel burnt. This point of injection at 65% stoichiometric will be referred to as the point of combustion deterioration.

In the engine illustrated at figure 2 the means of injection is conventional. T.D.C indicates Top Dead Centre in a cylinder of a reciprocating engine. Other angles being given in relation to T.D.C. "A" denotes main injection of fuel and is indicated by vertical hatchings and in figures 3, 4,5,6 and 7 a secondary introduction of fuel is indicated by horizontal hatchings STO is used as an abbreviation for stoichiometric. In each set of diagrams the far left hand diagram "A" illustrates low power (tick over) and for the right diagram "C" or "D" indicates full

power. In each case diagram "B" indicates a transition point.

In the conventional compression ignition engine using diesel fuel illustrated in figure 2 there is a single main injection of fuel which usually commences at or about 20 degrees before T.D.C for a direct injection engine or about 10 degrees before T.D.C for an indirect compression ignition engine.

At this point the cylinder is hot enough to initiate ignition of the injected fuel and the combustion starts shortly after injection commences and finishes shortly after injection ceases. at full power injection may continue up to or beyond T.D.C. Although 70% STO tends to be a maximum value higher values causing a smoky exhaust. The diagrams are drawn to illustrate progressively longer periods of injection from low to high power consistent with an injector construction and size wherein the rate of injection being varied by varying the period of operation of the injector. This is in accordance with conventional practice but it will be appreciated that with appropriate re-design of the injector system fuel quantity could be regulated by variation of the rate of supply instead of or as well as the period of injection.

A first preferred embodiment of an engine operating in accordance with the invention is identical with figure 2 engine over a lower part of its power range from "A" to "B", but the commencement of

the main injection is retarded by 5 degrees and commences at 15 degrees before T.D.C. This gives a quieter and smoother response at tick over and low load. The transition point "B" is chosen in relationship to the characteristics of the fuel, to occur at that position wherein after combustion of the main fuel portion the temperature within the cylinder is everywhere greater than 1150 degrees. This is usually at a position wherein the quantity of the fuel injected in the main injection is about 25% - 45% of the usual 35% STO. It is difficult to specify the transition point until the compression ratio and the initial amount of secondary fuel injected is taken into consideration. When these facts are known it is comparatively easy to select the correct transition point for the different types of engine.

Above the transition point "B" up to a maximum power (diagram C) the quantity of the main fuel injected is progressively increased up to a point just before the onset of combustion deterioration. This corresponds with a fuel injection rate of 65% STO.

This increase is less than would occur with the conventional engine (fig.2). From the point "B" upwards in the power range an additional injection of the same fuel takes place before the main injection and with air in the cylinder forms a mixture which is ignited upon ignition of the main injected fuel.

The engine in figure 3 uses an additional fuel injection from the

same injector as supplies the main fuel charge. This can be achieved by a modified jerk pump as described in accordance to a method of the invention and illustrated in figure 8 or an electronic control of the injection arrangements. The quantity of the second charge is held at about 5% STO. Because the secondary charge is pre-mixed, it is possible to achieve full power that consists of 65% STO main and 5% STO secondary fuel to achieve a 70% STO fuel injection as in figure 2 without the onset of combustion deterioration.

The secondary injection of the first preferred embodiment is timed to coincide with the commencement of the pressure stroke. When atomized secondary fuel is injected into the engine early in the compression stroke it is more concentrated than would be the case when injected earlier in the induction stroke when more time is available for mixing with the air entering the cylinder.

Although the fuel is more concentrated it is too lean to cause premature ignition during the compression stroke but there is a higher rate of thermo-chemical heating (pre-flame reaction).

This release of extra heat makes up for the reduction in temperature caused by the main injected fuel under maximum acceleration and will advance ignition timing by about 5 degrees to match the advance necessary to compensate for increasing speed and power.

The injection sequence described is not to be confused with early pilot injection. This method of operation injects a pilot charge of fuel early in the compression stroke from tick over to full load. Although this mode of operation reduces noise it takes no account of the transition point or timing of the main injection. At low load it will increase hydro-carbon carry over and at high load it will start the main fuel ignition too early resulting in some loss of power and reduced engine efficiency.

In the engine illustrated in figure 4 the same procedure applies as in figure 3 but both the timing of the main injection and the quantity of secondary fuel added are fully modulated by appropriate engine management. At tick over diagram "A" to reduce noise and vibration, the main injection is retarded to 10 degrees before T.D.C. Under acceleration injection advances to 15 degrees before T.D.C until the transition point is reached (diagram B).

At this point secondary fuel is introduced into the engine as an additional charge by the same injector during the induction stroke. The secondary injector of fuel is varied in proportion to engine load and may start at say 1% STO at the transition point "B" , up to 10% STO at point "C" and 20% STO at maximum load point "D".

At the same time the main injection of fuel is progressively

increased up to a maximum of 60% STO at full load point "D".

The main injection timing is retarded in proportion to the secondary fuel added to coincide with the generation of pressure rise in the combustion chamber at T.D.C.

In figure 4 diagram "A" illustrates tick over, diagram "B" the transition point, diagram "C" injection of 50% STO main fuel and 10% STO secondary fuel, diagram "D" is full power with 60% STO main fuel and 20% secondary fuel. The diagram also denotes a main injection timing of 10 degrees before T.D.C at tick over "A", 15 degrees before T.D.C at "B" , 10 degrees before T.D.C at "C" and 5 degrees before T.D.C at "D", maximum load. This gives a total of 80% STO fuel at full power, without combustion deterioration.

Figure 5 illustrates a variation of the method illustrated at figure 3 and 4 in which high boiling point fuel (diesel) is injected via a secondary injection on to a hot surface in the inlet manifold during an induction stroke of the engine. Such surfaces are described in the method of the invention illustrated at figure 9 and 10. At the point of impingements instant vaporisation of the fuel takes place causing complete mixing with the air being drawn in by the engine.

"A" in figure 5 denotes tick over where main injection is

retarded by 5 degrees and "B" the transition point. Beyond the transition point 0% STO to 5% STO is added in proportion to the load up to maximum power at "C". The main fuel is also varied in proportion to the load until it reaches 65% STO main injection plus 5% STO secondary fuel. This achieves full power without deterioration of the exhaust gases.

Figure 6 illustrates the same method of operation but at the transition point "B" the amount of secondary fuel added is variable from 0% to 10% STO at "C" full load. Because of the extra amount of secondary fuel added main injection timing must be retarded in proportion to the load.

That is 5 degrees at point "B" and 10 degrees at point "C", because of the extra heat generated during the compression stroke. This method of operation allows more power to be generated by the addition of extra fuel, 75% STO at full power "C" without exhaust pollution.

Figure 7 illustrates another preferred method of the invention, that is more suitable to the operation of the indirect compression ignition engine. In the indirect engine the combustion chamber is formed in the cylinder head and is attached to the main cylinder by a passage through which the air is accelerated by piston movement during the compression stroke. In this design of engine the compression ratio is increased to

about 22-1. To overcome the resistance of the passes through which the air is accelerated and also impart the degree of swirl necessary for good combustion. Because the combustion chamber is removed from the main cylinder it is comparatively easy to insulate. The additional heat of compression and the retention of heat by this method of operation reduces ignition delay to a minimum, so that main fuel injection is in the order of about 10 degrees before T.D.C.

The disadvantage of the indirect engine is that only about 50% of the compressed air volume of the cylinder can be forced into the pre-combustion chamber. This method of operated gives good combustion at low and medium load because of the high rate of mixing caused by the high speed swirl, but is less efficient at high load. Because combustion conditions become too rich in the pre-mixed chamber the speed of flame propagation is reduced.

In figure 7 the main injection is regarded to 5 degrees before T.D.C at "A" and the preferred method would be to advance injection timing as the load increased until the transition point illustrated at "B" is reached. Thereafter secondary fuel would be added in proportion to the load up to a maximum of 20% STO. At the same time main injected fuel would be added in proportion to the load until a maximum of 60% STO is reached.

This method of operation would coincide with 50% STO main fuel

and 10% STO secondary fuel at point "C" and 60% STO main fuel injection and 20% STO secondary fuel at maximum power and load at point "D". Main injection timing would be retarded to 5 degrees at point "C" and 3 degrees before T.D.C at point "D".

It will be understood that the conditions described are by way of example only and can be varied depending on the compression ratio or the addition of turbo-charging so that injection timing and quantity of secondary fuel added are infinitely variable and need to be selected for different engine operating conditions.

Although the methods described are capable of reaching the standards of exhaust emission required by law it may be necessary with future legislation to control main injection timing and secondary fuel quantity so that the amount of nitrous-oxide gas from the exhaust can be substantially reduced.

Because the generation of nox is related to temperature and maximum pressure in the cylinder it may require the main injection to start at T.D.C, this would continue until the transition point is reached, beyond this point main fuel injection would be further retarded in proportion to the amount of secondary fuel added.

I have described in my previous patent a method of introducing secondary fuel to the engine during the induction stroke by

impinging the fuel on a hot surface such as the inlet valves but a more simplified method will now be described and is illustrated in figure 8.

it is common practice to introduce a portion of exhaust gas into the inlet manifold of the diesel engine to reduce nitrous oxides (nox) concentration and is referred to as EGR exhaust gas recirculation.

A duct is provided to connect the exhaust manifold to the inlet manifold through which exhaust gas can be drawn to be mixed with the intake air. There is a valve provided in the ducting which regulates the flow of hot exhaust gas in proportion to engine requirements. A fuel injector can be positioned as shown in diagram 9 to impinge fuel onto the valve surface. On impingement the fuel breaks up mixed with the hot exhaust gas flowing through the ducting.

Although the exhaust gas will be over the temperature required for spontaneous ignition, no ignition takes place because there is insufficient oxygen remaining in the exhaust gas to support combustion. Although there is no ignition the high temperature causes the fuel to be instantaneously gasefied. Because the portion added is small in relationship to intake air volume the gasefield fuel will be rapidly cooled when it enters the inlet manifold, thereby eliminating any possibility of premature

ignition.

Another method for the introduction of secondary fuel according to a method of the invention is illustrated at figure 9. A glow plug which is in common use for the pre-heating of air in the indirect engine to assist in starting is positioned to include into the inlet manifold.

An injector is angled in a position above the glow plug so that when a stream of fuel is discharged from the injector tip it will impinge on the glow plug surface. By impinging fuel off the hot plug surface instantaneous vaporization takes place. This causes the vaporized fuel to be entrained and mixed with the induction air as it passes through the manifold. The method described are by way of example only or may be varied in different ways.

Although secondary fuel can be added from a source remote from the engine as a gas, propane, butane, methane, etc. or vaporized petrol, paraffin or diesel and the quantity varied by controlling the air-flow to the inlet duct (such a device could be a carburetter). Unless the transition point is taken into consideration the fuel will pass through the engine and will be released with the exhaust gas as partially burnt hydro-carbons.

Although petrol has a high resistance to pre-flame reaction there will be considerable heat release during the compression stroke.

This will advance the onset of ignition of the main charge causing a rise in pressure before T.D.C which will have an adverse effect on the engine efficiency.

When paraffin or diesel vapour which has a low resistance to pre-flame reaction is added to the inlet air supply the heat generated by compression will cause the main fuel charge to be ignited as soon as it leaves the injector. This will cause a big drop in power and increased fuel consumption. Also over a period of operation may cause considerable damage to the engine.

A device according to WO/92/05360 fitted to an engine without taking these requirements into consideration gave the following results:-

Device not in use

(Fuel injection at TDC-7.5°.)

M.P.H	B.H.P
40	36
50	56

Acceleration 30mph to 50mph in top gear 29.8 sec.

30.3 " .Average 30.05.

Engine at working temperature.

<u>Device in use</u>

(Fuel injection TDC - 7.5°.)

M.P.H	B.H.P
40	34
50	50

Acceleration 31.9 sec.

28.8 " Average 30.35

Engine at working temperature.

Fuel consumption = down by 1.2% to 2.0%

<u>Device re-adjusted by J.H Greenhough to my patent specification</u>

(Injection at TDC-2.5%).

M.P.H	B.H.P.	Improvement	
		From Device Makers Standard	From Device Makers Standard
40	40	17.64%	11.1%
50	62	24%	10.7%

Time to Acceleration from 30-50 mph 28.2 sec. (cold)

" " " " 24.9 " (hot)

Improvement 20% when hot.

The invention can be applied to a four stroke engine, a two-stroke engine or a rotary engine.

Many variations are possible within the scope of the invention.

Claims

1. A method of operating a compression ignition internal combustion engine wherein over a lower part of the power range there of from tick over up to a transition point, engine power is varied by varying the quantity of the main injection late in the compression stroke from a minimum at tick over up to a higher value at the transition point and there after, over a higher part of the power ranges from the transition point up to full power, the main injection of fuel is supplemented by a secondary introduction of the same fuel, wherein the main injection is delayed when secondary introduction takes place.

2. A method as claimed in claim 1 in which the delay is fixed over the higher power range.

3. A method as claimed in claim 1 or claim 2 in which the secondary introduction of fuel prior to the main injection of fuel is accompanied by a retarding of the main injection.

4. A compression ignition engine having means for introducing a secondary charge of fuel prior to a main fuel injection over a higher power range of the engine and means for retarding said main injection during said higher power range.

5. An engine as claimed in claim 4 in which the introduction of fuel is by injection, by carburettor by inclusion into the induced air or by introduction into recycled exhaust gas or otherwise.

6. An engine as claimed in claim 4 or claim 5 in which the amount of retardation is varied dependent upon the amount of secondary introduction.

7. An engine as claimed in claim 4, claim 5 or claim 6 in which the retardation starts from a relatively large value tick over, goes to a lower value at transition and thence to a larger value at full power.

8. A method of operating a compression ignition internal combustion engine substantially as herein described.

9. A compression ignition engine constructed and arranged for use and operation substantially as herein described.

Patents Act 1977 **Examiner's report to the Comptroller under Section 17** **(The Search report)** -30-	Application number GB 9307696.6
Relevant Technical Fields (i) UK Cl (Ed.M) F1B (ii) Int Cl (Ed.5) F02B 7/00, 7/02, 7/04, 7/06, 7/08, 75/12	Search Examiner R J DENNIS Date of completion of Search 10 MAY 1994
Databases (see below) (i) UK Patent Office collections of GB, EP, WO and US patent specifications. (ii)	Documents considered relevant following a search in respect of Claims :- 1 to 9

Categories of documents

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&: Member of the same patent family; corresponding document.

Category	Identity of document and relevant passages		Relevant to claim(s)
Y	GB 2169960 A	(GREENHOUGH)	4 to 7
X,Y	GB 0878278	(INSTITUT) see particularly lines 88 to 97, page 5	X:1 to 3 Y:4 to 7
Y	GB 0729383	(TEXACO)	4 to 7
Y	GB 0714672	(STANDARD)	4 to 7

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